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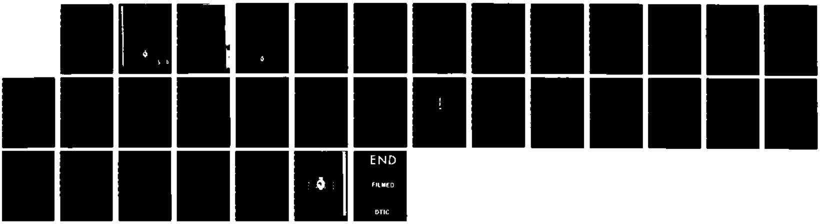

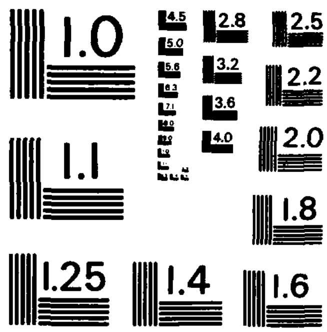

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ANALYSIS OF THE MACKENZIE CLASS MAST OF HMCS QU'APPELLE

Neil G. Pegg - Donald R. Smith
Donald J. Hussey

Defence Research Establishment Atlantic

Centre de Recherches pour la Défense Atlantique

Canada

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9 GROVE STREET
P.O. BOX 1012
DARTMOUTH, N.S.
B2Y 3Z7

TELEPHONE
(902) 426-3100

CENTRE DE RECHERCHES POUR LA DÉFENSE ATLANTIQUE
9 GROVE STREET
C.P. 1012
DARTMOUTH, N.É.
B2Y 3Z7

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Approved by B.F. Peters A/Director/Technology Division

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ABSTRACT

The finite element program VAST, and experimental vibration data are used to analyze a serious vibration problem with the new mast on the Mackenzie class ship, DDE 261, HMCS QU'APPELLE. It has been determined that the mast's natural frequency of vibration falls into the range of excitation frequencies produced by the propeller shaft rotation. It is postulated that the propeller excitation is driving both resonant hull and mast modes together.

After analysis of some possible solutions using VAST, it is determined that there is no simple solution that would shift the mast's natural frequency completely out of the range of excitation resulting from the propeller shaft rotation. However, some trial solutions are suggested that may provide sufficient attenuation of the mast vibration for safe and normal operation of the ship. In addition, an analysis approach is suggested which would avoid this type of problem in future designs.

[illegible]

Résumé

Le programme à éléments finis VAST et des données expérimentales relatives aux vibrations sont utilisés pour analyser un grave problème de vibration qui concerne le nouveau mât du HMCS QU'APPELLE, un navire de la classe Mackenzie portant le matricule DDE 261. Il a été constaté que la fréquence naturelle de vibration du mât se trouve dans la gamme des fréquences d'excitation produites par la rotation de l'arbre de l'hélice. On suppose que l'excitation produite par l'hélice amplifie à la fois les fréquences de résonance de la coque et celles du mât.

Après analyse de certaines possibilités de solutions à l'aide de VAST, il a été constaté qu'il n'existe pas de solution simple qui permettrait de placer la fréquence de résonance naturelle du mât complètement à l'extérieur de la gamme des fréquences d'excitation produites par la rotation de l'arbre de l'hélice. Toutefois, certaines solutions provisoires sont proposées; elles fourniront peut-être une atténuation suffisante de la vibration du mât pour permettre une exploitation sûre et normale du navire. En outre, l'auteur propose une méthode d'analyse qui permettrait d'éviter ce type de problème à l'avenir.

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1. INTRODUCTION

The mast of the Mackenzie Class destroyers, shown in Figure 1, was recently modified to accomodate new equipment, primarily a UHF polemast antenna assembly to improve its communication capability. The details of the modifications are given in Reference 1. The engineering was carried out by Martec Ltd under contract to Director of Maritime Engineering and Maintenance (DMEM)[2]. Under this contract the mast was analized and designed for its ability to withstand shock, ship motion, air blast and aerodynamic loading. Its first natural frequency of vibration was estimated. The contract did not ask for a full frequency analysis.

The first of the new masts was fitted on HMCS QU'APPELLE, stationed at CFB Esquimalt, British Columbia. During initial trials after the refit period the mast was found to vibrate in such a manner as to cause the flag deck to be declared 'out of bounds' in the interests of crew safety.

The Defence Research Establishment Atlantic (DREA) agreed, under tasking from DMEM, to investigate the cause of, and propose a solution to, the vibration problem.

Initially it was decided to carry out an analysis of the mast using the finite element program VAST[3] to determine the mast's natural frequencies of vibration. The data for the original finite element model of QU'APPELLE's mast, used in the design process in 1981, was obtained from Martec and converted to run with VAST. The VAST vibration analysis suggested that two of the mast's natural frequencies occurred within the range of excitation frequencies produced by the propeller shaft rotation and that a resonance condition with the propeller rotation might exist. The shaft rate at which the mast was observed to vibrate excessively by the crew of QU'APPELLE was 156 rpm or 2.6 Hz.

To determine if the shaft rotation frequency was indeed the cause of the mast vibration problem on HMCS QU'APPELLE, it was decided that vibration tests should be performed on the mast to determine its exact natural frequencies. Natural frequencies determined from mathematical models such as in finite element analysis are generally higher than the actual values. This is caused by idealization of fully constrained connections and boundary conditions which are in reality somewhat flexible. Thus field trials are often necessary to determine exact values of natural frequencies to verify the theoretical results and to pinpoint the cause of the vibration. It is noted here that the value of the first natural frequency determined using VAST for a finite element analysis was 6% higher than the measured value; this is quite good agreement for a mathematical model.

Vibration trials were first performed on the mast of HMCS SAGUENAY alongside at Halifax. HMCS SAGUENAY has a similar mast to that of HMCS QU'APPELLE although there are some structural differences. These trials were used to evaluate the proposed testing method and check equipment before embarking for Esquimalt. Having been proved successful in the SAGUENAY tests, the same methods were then applied to QU'APPELLE alongside at Esquimalt. The resulting data were analyzed at DREA to determine the mast's natural frequencies of vibration and damping characteristics. The results of the SAGUENAY tests also proved useful as a comparison to the QU'APPELLE results and are given in this report for that purpose.

In order to perform a complete dynamic analysis of the mast structure, and of proposed modifications to it, the total spectrum of input vibration sources to the mast needed to be determined. On request from DREA, Defence Research Establishment Pacific (DREP) performed sea trials to establish the characteristic vibrations being transmitted to the mast as the ship was accelerated through its speed range. These tests could not be performed by DREA at the time of the original vibration trials as HMCS QU'APPELLE was not operational.

Once the existing mast's characteristics were determined and the nature of the exciting forces which acted on it were evaluated, design modifications to correct the excessive vibration problem were investigated. The finite element program VAST was the major tool used to analyze the effects of various structural modifications on the dynamic characteristics of the mast.

Vibration problems are by no means new to naval structures. In reviewing design standards for masts, it was found that analyses to prevent resonance conditions are a standard design requirement. The United States Navy mast design standard[4] and the Engineering Standard for the Design of Lattice Masts of Ships[5] developed by Martec Ltd for DMEM, both specify that the mast natural frequencies should be determined and checked to insure that they will not result in a resonant condition with possible excitation sources from the ship. These possible sources of excitation consist of[4]:

1) vibration from propeller shaft rotation,
2) vibration from propeller blade rate (shaft rate multiplied by the number of blades on one propeller),
3) vibration induced by rotating radar antennae, and
4) motion induced by the hull natural modes of vibration.

Of particular concern to the excitation of lattice type mast structures are the propeller shaft rotation rate and the hull mode vibration. The frequency range of excitation produced by these sources is 1.0 to 4.0 Hz. Lattice masts and polemasts typically have natural frequency values in this range. A serious condition of resonance can result if the mast and hull natural frequencies are the same and are both excited by shaft rotation.

This report discusses the structural analysis, field trials and the possible solutions to attenuate the mast vibration. In addition, suggestions are made as to how this problem could be avoided in future mast design work.

2. CHARACTERIZATION OF THE EXCITATION MECHANISM

2.1 Method

The DREA approach to the analysis of the mast vibration problem involved several steps. The first was to use VAST to perform an analysis on a finite element model of the mast to estimate its dynamic characteristics. The second step was to perform vibration tests on the mast to verify the finite element model and to determine exactly the dynamic characteristics of the mast. The third step was to try to determine the excitation functions acting on the mast. Once the excitaion mechanism was identified various corrective procedures could be evaluated; varying from minor mast modification to complete redesign.

2.2 Finite Element Model

The finite element model of HMCS QU'APPELLE's mast is shown in Figures 2 and 3. This model was generated from data supplied to DREA by Martec Ltd. The model consisting of beam and membrane elements is quite complex: over 200 elements are used. The structural weight is incorporated in the model by supplying material density to the members; additional equipment weight is included using lumped masses attached to the structure.

A number of differences between the finite element model and the actual mast were discovered at the outset of the investigation. These included some plate thicknesses which were too large in the model and some lightening holes of significant size which were completely plated over. This made the model stiffer than the actual mast. The model was corrected prior to the VAST analysis. It is most probable that some of the weights and their locations supplied for the original 1981 analysis were not exactly the same as those actually attached to the mast in 1983. In consideration of this, VAST was not to be expected to give exact values of the natural frequencies of vibration.

2.3 Mast Vibration Trials

Initial frequency analysis with VAST indicated that the mast had a fundamental natural frequency of around 2.75 Hz (Figure 4). Additional modes of vibration are shown in Figures 5 and 6. The 2.75 Hz value was sufficiently close to the 2.6 Hz critical shaft rate frequency (the shaft rate at which QU'APPELLE's crew observed the large vibrations in the mast) to warrant field trials on the mast to determine its exact natural frequencies. Once known, these frequencies could be used to determine accurately the cause of the excessive vibration.

The mast structure (Figure 1) is quite large and complex. It supports very sensitive and expensive equipment. Therefore, a method for evaluating the mast natural frequencies was chosen that would be simple to use and would not cause unduly large motions of the mast with the possibility of damage to some of the equipment. The mast was excited into free vibration by two methods:

1) manually shaking the mast by jerking it at the second platform level (Figure 1), and

2) striking the mast at the bottom of the lattice structure with a hammer (Figure 1).

The first method set the mast primarily into its first mode of vibration. It was noted that it was quite easy to cause large displacements of the order of 30 centimeters peak to peak at the top of the polemast. The second method tended to excite the structure into several modes of vibration and was used to obtain some of the higher natural frequencies of the mast. The mast motions were measured using accelerometers. These were placed in pairs to measure motions in both the fore-aft and athwartship directions (longitudinal and transverse axis of the ship). The accelerometers were placed at five stations shown in Figure 7:

1) top of the polemast in way of the aircraft warning lights,
2) top of lattice mast,
3) top (first) platform level,
4) second platform level, and
5) near the bottom of the mast structure where it is attached to the flag deck (01 deck).

Since only six accelerometers were available (making three pairs), all levels could not be tested at the same time. Also the mast was only excited in one of the two directions for each test. Therefore there were eight separate trials performed. These are described in Table 1.

The accelerometer outputs were recorded on six channels using a Hewlett Packard 3968A Instrumentation Analogue Tape Recorder. A Hewlett Packard 3582A Spectrum Analyzer was also employed to give immediate natural frequencies of the system while the tests were being performed. The equipment was set up on the bridge of the ship to protect it from the weather, and cables were run out to the accelerometers on the mast.

As discussed in the introduction, vibration trials were performed first on HMCS SAGUENAY in Halifax to test the method and equipment. Only one accelerometer was used in these tests. Results of both of these trials are given in Tables 2 and 3 for comparison.

2.4 Analysis of Excitation Source

The four most likely sources causing vibration in a mast structure are listed in the introduction (as taken from Reference 4). The two causes of concern in this case were the propeller shaft rotation rate and the hull modes of vibration. The radar rotation frequencies were too low and the blade rate frequencies were too high to cause a resonance problem with this particular mast. The vibration frequencies resulting from the shaft rate of QU'APPELLE range from 2.2 to 3.8 Hz.

The values of the hull mode frequencies for QU'APPELLE are not currently known. The hull modes could be measured experimentally or estimated by creating a finite element or beam model of the entire ship. This would be a fairly involved process, but would be worthwhile for more modern and future ships such as the DDH 280 and the CPF. The first hull modes of naval ships are generally found in the range of 1.0 to 4.0 Hz. This creates the possibility that the mast's natural frequency may match a hull frequency on QU'APPELLE, and result in a double amplification of the mast vibration when the ship is operated at 156 rpm. If such were the case, a small shift in the natural frequency of the mast would move it off of the hull mode and the amplification would be greatly reduced.

In order to determine the nature of the exciting force to the mast, it was decided to make measurements at the base of the mast while the ship went through its speed runs. These trials were carried out by Defence Research Establishment Pacific. Accelerometers were mounted on platform 4 (Figure 1) at the top of the shell casing part of the mast. Records were made as the ship went through speed run trials from 110 to 170 rpm. Bad weather prevented the ship from achieving higher speeds.

3. MAST MODIFICATIONS USING VAST

Analysis of various structural modifications to shift the mast's natural frequency of vibration away from possible excitation frequency sources was carried out using the finite element program VAST[3]. The alternatives involved either adding mass to the structure to lower its natural frequency or stiffening it to raise the natural frequency. Methods of adding damping to reduce the amplitudes of vibration were also considered. The criteria determined by DREA in choosing possible modification alternatives were as follows:

1) to be able to make the necessary changes to the mast without removing it from the ship,

2) to keep the present geometry intact so as not to cause any interference with electronic sensing gear, and

3) to keep the changes as easy to implement as possible by avoiding excessive welding and plate cutting.

The best solution to maintain the structural integrity in all load cases, although not the easiest to implement, was to stiffen the structure and increase its natural frequency. This could be achieved by adding some additional members and increasing the size of some existing members in the model. Some modifications investigated also involved using steel instead of aluminum. The modifications analyzed to stiffen the mast are shown in Figure 8 and are as follows:

1) addition of a sleeve over the bottom portion of the pole mast,

2) addition of two vertical 127x127 mm (5x5 in) angles between platforms 1 and 3 parallel to the aft lattice verticals,

3) addition of cross bracing between verticals running from platforms 1 to 2,

4) increasing the strength of the lattice mast by adding half sections of 89 mm (3.5 in) extra strong pipe to the existing 73 mm (2 7/8 in) vertical pipe legs of the lattice mast, analysis was performed using both aluminum and steel half sections, and

5) replacement of existing aluminum polemast with a stiffer composite material polemast.

The stiffening of the structure between platforms 1 and 2 was felt necessary as it was noted in reference 1 that this was the weakest area of the structure. It was also observed by DREA personel during the vibration trials to be a location of possible structural weakness.

Modification 4 could be implemented by clamping the half pipe to the existing vertical pipe sections on the lattice mast. In order to determine the feasibility and actual effect on stiffness, sections of the 89 mm and existing 73 mm aluminum pipe were clamped and load tested at DREA. Results of this test proved this solution to be feasible.

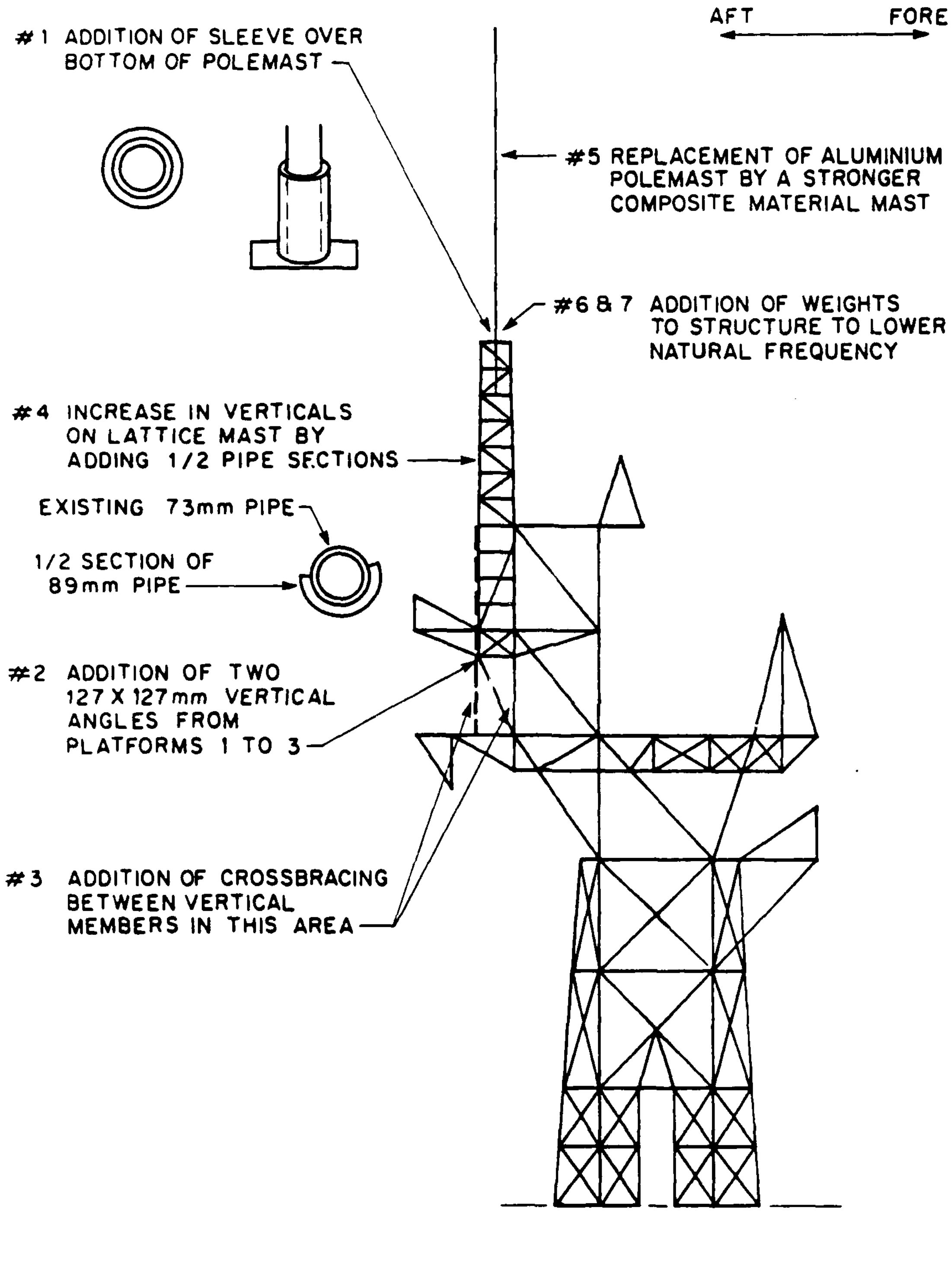

FIGURE 8: PROPOSED MODIFICATIONS TO QU'APPELLE MAST

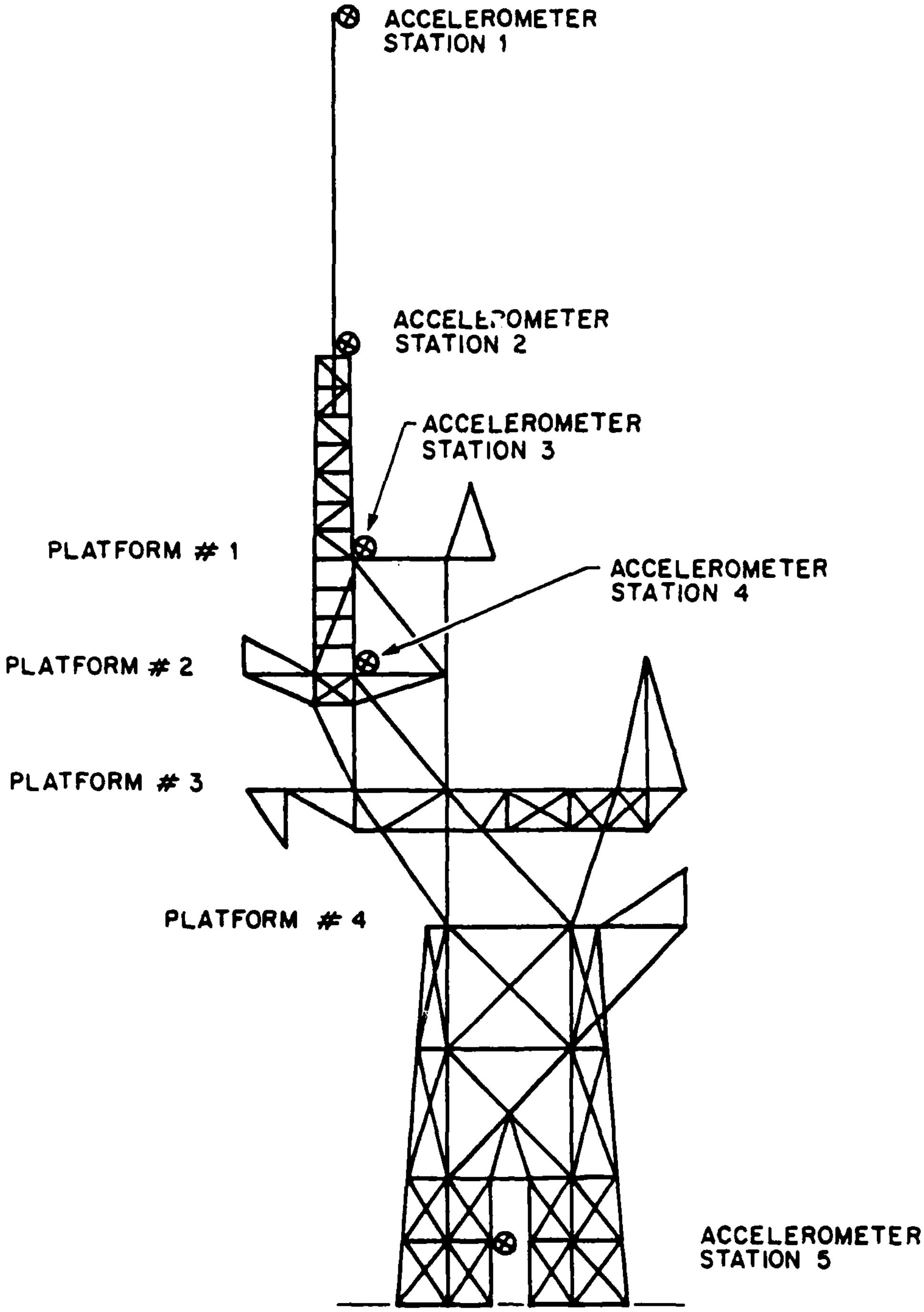

FIGURE 7: ACCELEROMETER POSITIONS FOR VIBRATION TRIALS

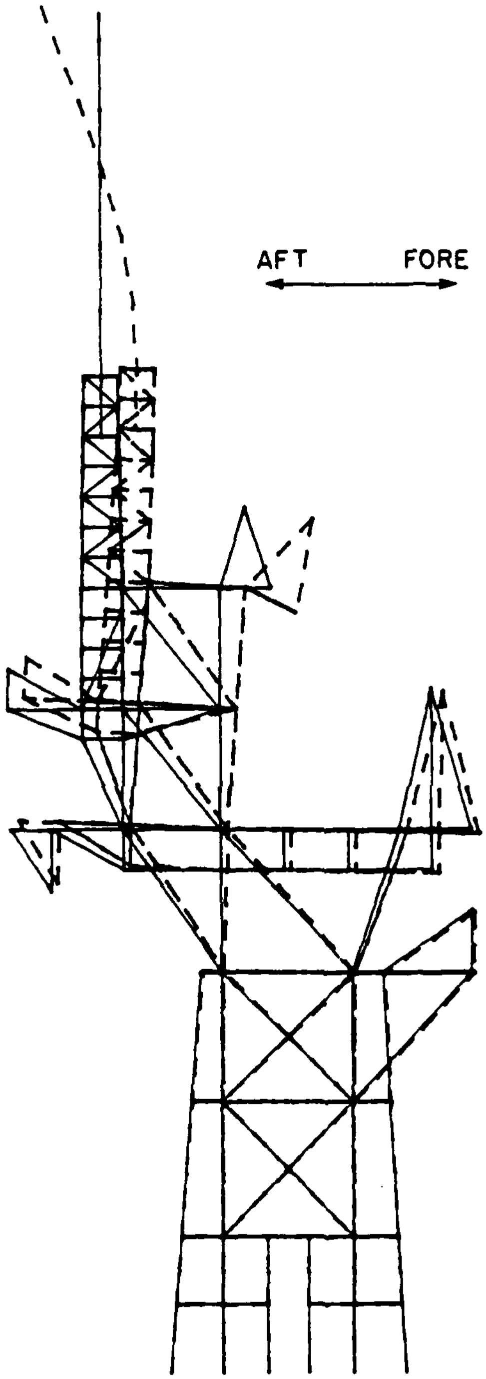

FIGURE 6: SECOND LONGITUDINAL MODE OF VIBRATION 4.1 Hz

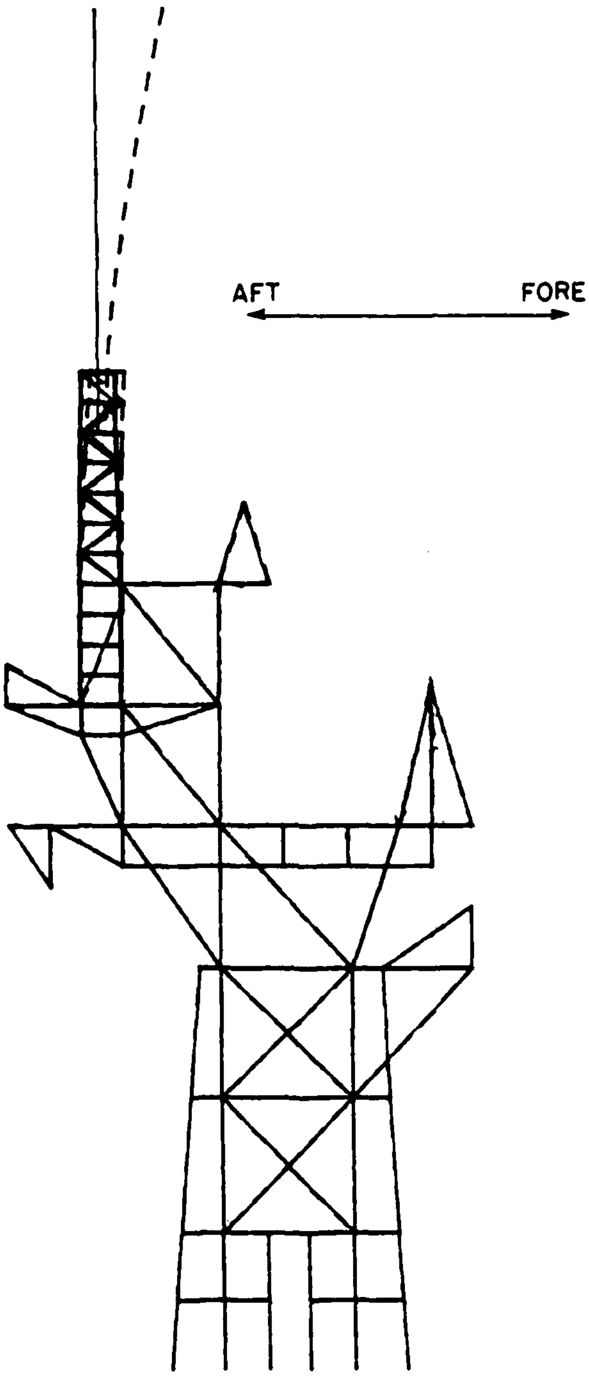

FIGURE 5: FIRST LONGITUDINAL MODE OF VIBRATION 2.52 Hz

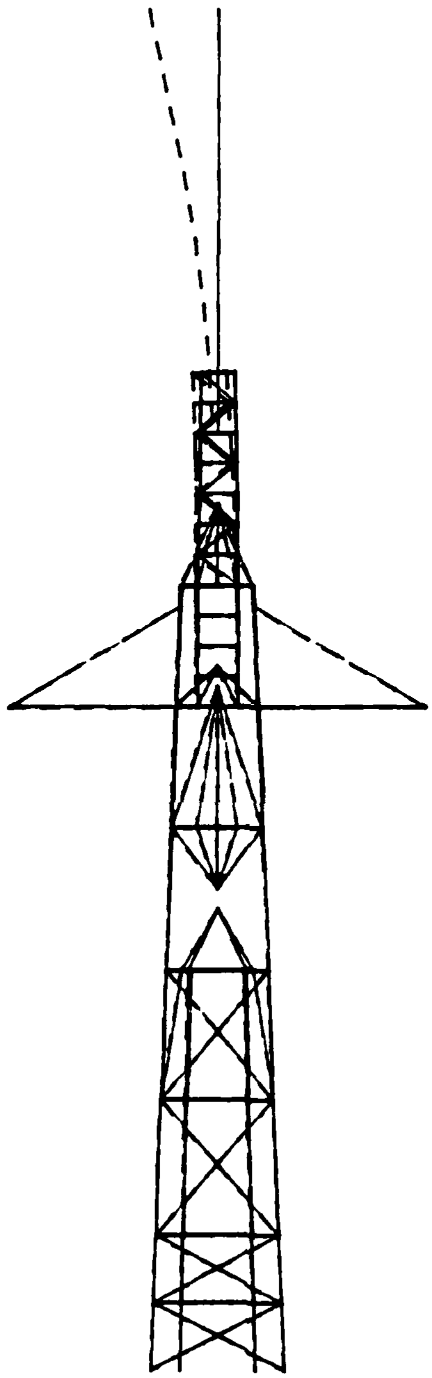

FIGURE 4: FIRST TRANSVERSE MODE OF VIBRATIONS
VAST = 2.75 Hz EXPERIMENT = 2.6 Hz

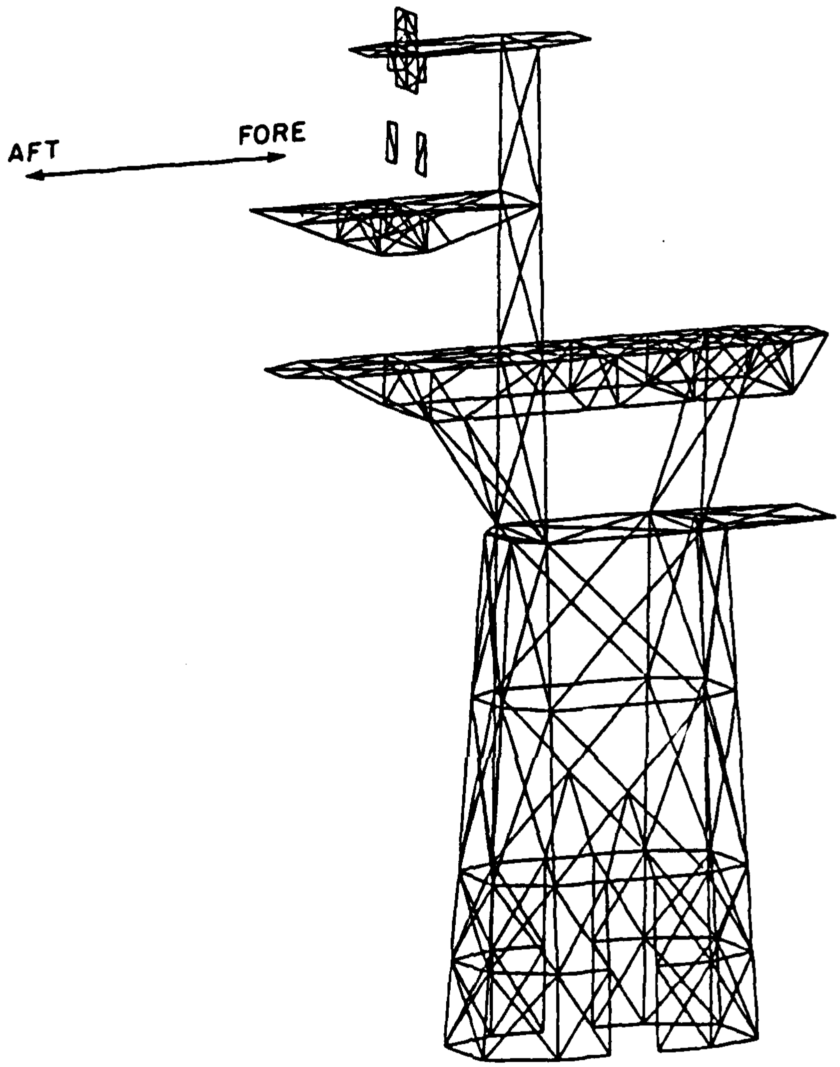

FIGURE 3: FINITE ELEMENT MODEL; PLATE ELEMENTS

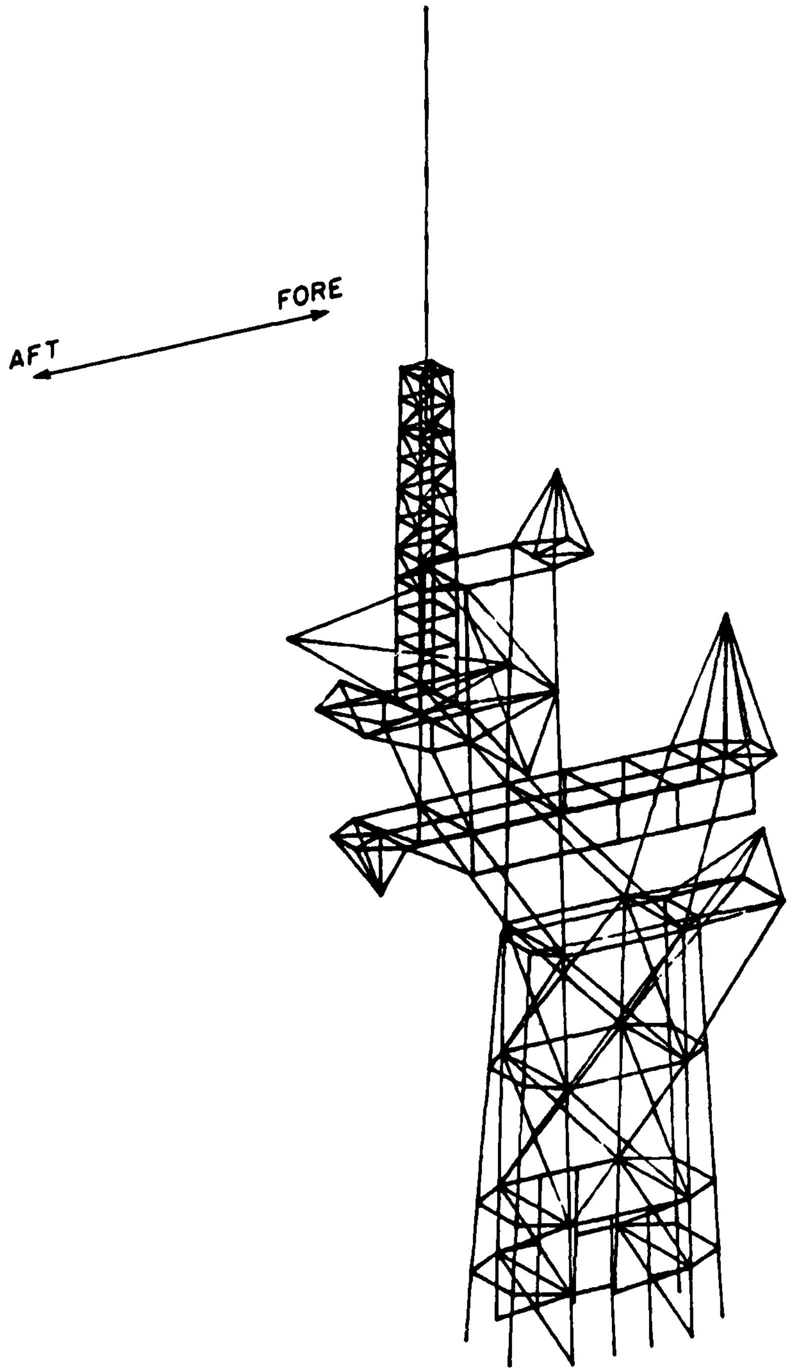

FIGURE 2: FINITE ELEMENT MODEL; BEAM ELEMENTS

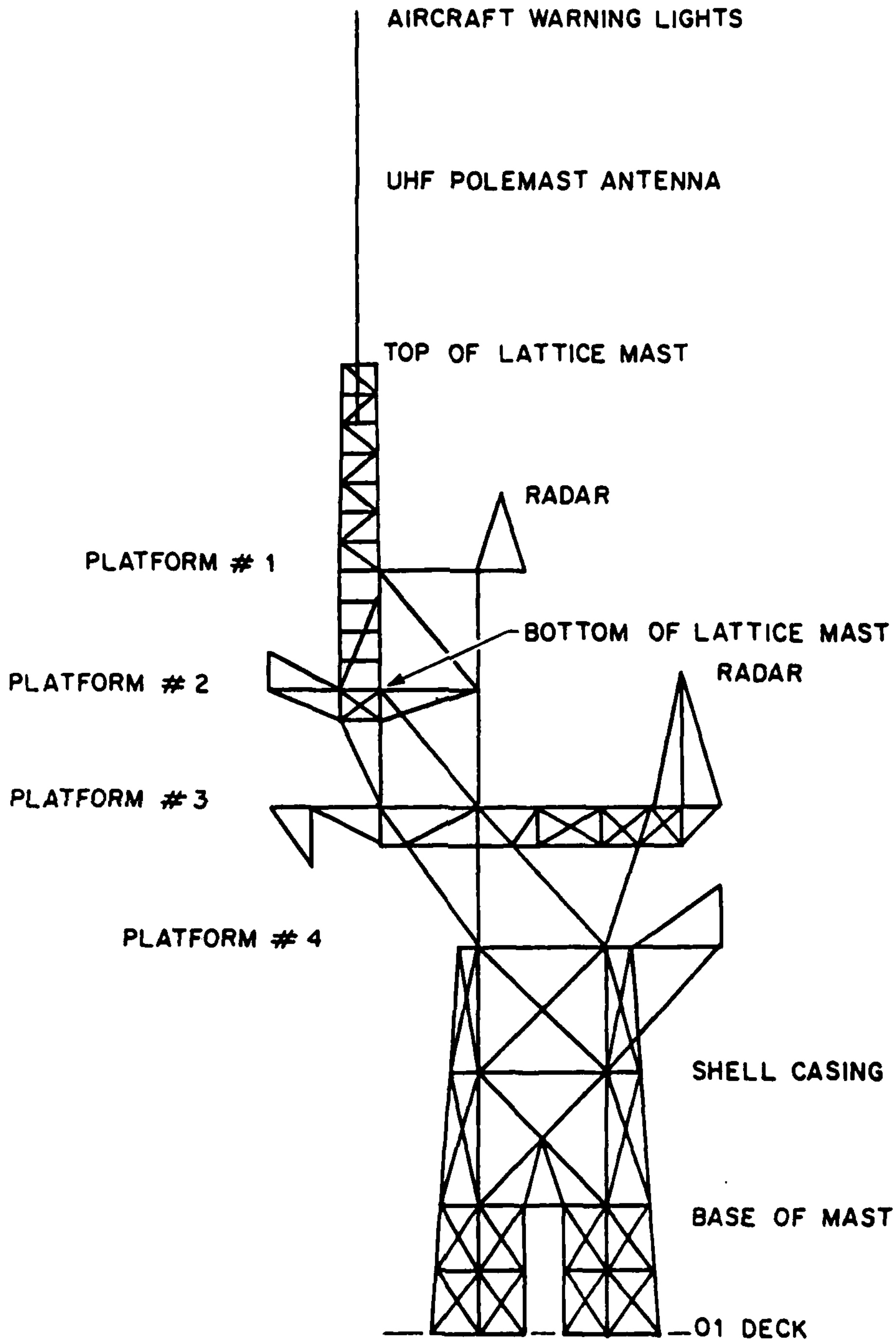

FIGURE 1: MACKENZIE CLASS MAST WITH POLEMAST MODIFICATION

TABLE 4

TRIAL MODIFICATIONS TO THE MAST USING
THE VAST FINITE ELEMENT PROGRAM

TRIAL	MODIFICATION	CHANGE IN FUNDAMENTAL FREQUENCY (HZ) FORE/AFT	ATHWARTSHIP
1	Addition of sleeve over bottom portion of polemast	+ 0.3	+ 0.39
2	Addition of vertical 127 mm angles between platforms 1 and 3	+ 0.06	+ 0.04
3	Addition of cross bracing between verticals at platforms 1 and 2	+ 0.02	+ 0.02
4	Adding half 89 mm pipe sections to existing 73 mm vertical sections of lattice mast -aluminum	+ 0.32	+ 0.29
	-steel	+ 0.33	+ 0.30
5	Replacement of existing aluminum polemast with a composite mast *	+ +	
6	Addition of 90 kg weight at the top of the lattice mast	- 0.07	- 0.10
7	Addition of 180 kg weight at the top of the lattice mast	- 0.11	- 0.13
	Combination of 2,3 and 4 using steel	+ 0.8	+ 0.9

* increase would depend on material properties of composite

TABLE 3

TABLE OF MEASURED NATURAL FREQUENCIES OF THE MAST OF HMCS SAGUENAY (Hz)

FORE/AFT	2.40	3.96	4.40	6.60	8.90	9.30	13.00	19.60
PORT/STARBOARD	2.02	3.78	4.02	5.90	7.90	12.00		

TABLE 1

DESCRIPTION OF TRIALS

TRIAL	EXCITATION OF MAST	DIRECTION OF MOTION	ACCELEROMETER LOCATIONS (Fig.7) 1	2	3	4	5
1	Manually Shaken	F/A	X	X	X		
2	Manually Shaken	P/S	X	X	X		
3	Struck with Hammer	F/A	X	X	X		
4	Struck with Hammer	P/S	X	X	X		
5	Manually Shaken	F/A	X	X		X	
6	Manually Shaken	P/S	X	X		X	
7	Manually Shaken	F/A	X	X			X
8	Manually Shaken	P/S	X	X			X

TABLE 2

TABLE OF MEASURED NATURAL FREQUENCIES FOR THE MAST OF HMCS QU'APPELLE

TRIAL	STATION	DIRECTION	FREQUENCIES (HZ)							
1	1	F/A	2.52	4.1	5.2	7.5	9.1			
1	2	F/A	2.52	4.1	5.4	7.6	9.1	11.3		
1	3	F/A	2.52	4.1	5.4					
2	1	P/S	2.6	4.5	11.5					
2	2	P/S	2.6	4.5	6.1	11.5				
3	1	F/A	2.5	4.2	5.5	10.6	17.0			
4	1	P/S	2.6	4.5	10.3	11.5	20.6	23.3		
4	2	P/S	2.6	4.5	6.1	10.3	11.5	12.1	20.6	23.3

Once the exciting frequencies are determined, a careful analysis of the proposed structure using a finite element program such as VAST would be required to determine its natural frequencies of vibration. Some judgement on the part of the designer would be required to insure that the mast's natural frequencies fall into a suitable range outside that of possible vibration sources.

After completion of this report, in accordance with proposed modification 6 of Table 4, trials were performed using a 90 kg weight attached to the polemast at the top of the lattice structures (Figure 1). Observations made by personnel of HMCS QU'APPELLE and Naval Engineering Unit Pacific indicated that the mast's motion has been greatly reduced to within acceptable limits for safe operation. In view of this, it appears that shifting the natural frequency a small amount with the addition of the weight has successfuly solved the resonance problem. This indicates that the existance of a matched mast hull mode was quite likely.

The analysis of possible modifications to the mast using VAST indicated that a large change in its natural frequency is not feasible. The results of these modifications are shown in Table 4. Major structural redesign of the mast would be required to cause a large shift in its natural frequency. This result eliminated the possibility of shifting the mast's natural frequency out of the propeller shaft rate range (above 3.8 Hz) by implementing relatively simple modifications to the mast. As the loading function could not be accurately determined, there was no apparent solution that would guarantee attenuation of the mast vibration. However, DREA did suggest some possible solutions to DMEM which could be tried before a full redesign of the mast was deemed necessary. Two possible solutions were suggested and are given below.

Masts on similar ships, such as HMCS SAGUENAY (see Table 3), have natural frequencies in the same range as HMCS QU'APPELLE's mast, operate under much the same conditions, and do not have vibration problems. This indicates that there is something unique about the excitation force on QU'APPELLE's mast such as a matched mast-hull mode excitation. If the athwartship natural frequency is lowered by approximately 0.1 Hz by attaching a 90 kg weight to the top of the lattice mast, the result would be enough to greatly reduce the resonance problem. If adding weight does not prove to be successful, the alternative would be to stiffen the mast to raise its natural frequency. In shifting the natural frequency of the mast, there is always the possibility that it could end up in resonance with another shaft rate, possibly one which would impart even more energy to the mast and make vibrations worse. If the mast were stiffened by a combination of modifications 2,3 and 4 (shown in Table 4 and Figure 7) the natural frequency would increase by almost 1.0 Hz with a near doubling of the overall stiffness of the mast.

5. CONCLUDING REMARKS

As a result of this study, several recommendations for future mast design work are suggested. First, the possibility of the mast having a resonance condition driven by some exciting force on the ship should be considered. Existing DMEM design standards (references 6 and 7), make some reference to this problem, but do not go into much detail. To avoid any problems of this nature, the USN mast design standard (reference 4), suggests that the mast be designed so that its natural frequency is 25% greater than excitation sources from the hull modes or propeller shaft rotation rate. This is not always practical as other criteria for the mast design such as radar height and minimum profile area drive the design towards tall, slender structures supporting heavy equipment, giving low natural frequency values. In order to design for the condition where the mast's natural frequency cannot be placed outside the range of possible excitation, an accurate analysis of vibration sources in the ship should be performed. Some research should be done into developing the best analysis method for doing this.

To lower the frequency, lumped masses of 90 and 180 kg (200 and 400 pounds) were alternatively incorporated into the mast model at the top of the lattice (Figure 8). Although adding mass is more effective in changing the natural frequency if it is attached higher on the mast structure, heavy weights could not be easily attached to the structure at any point higher than the top of the lattice mast portion. It is noted here that attaching weights to the mast may induce other structural problems under shock or blast loading and that if this solution were chosen, additional analysis under the shock and blast loading conditions should be considered. Reference 1 indicates marginal safety factors for some members such as the 127x127 mm angles between platforms 2 and 3.

The effects on the natural frequency of the mast of all of the above modifications were evaluated using VAST and are summarized in Table 4.

The possibility of adding damping to the structure was also addressed. It was felt that in order for a damping device to be able to dissipate a significant amount of energy, the structure would have to undergo relatively large displacements which was the very problem that needed solving. It is also difficult to implement damping devices on a free standing cantilever structure. One possible alternative to induce energy dissipation into the system was the use of a mass damper. This device would consist of a mass, spring and damper unit of approximately the same natural frequency as the structure but of opposite phase in displacement. This alternative was given considerable attention; however, without an experimental phase of development with the actual mast structure it was felt that its performance could not be suitably predicted.

4. RESULTS AND DISCUSSION

Results of the alongside mast vibration trials conducted by DREA showed clear indication that the mast natural frequencies were 2.52 and 2.60 Hz in the fore-aft and athwartship directions respectively (Figures 4 and 5 and Table 2). The athwartship direction was the direction of the large motions reported by the crew of HMCS QU'APPELLE. The value of 2.60 Hz corresponded exactly with the shaft rate of 156 rpm (2.60 Hz) reported as the propeller revolution rate at which the vibrations were noticed. Figures 9,10,11 and 12 show Fourier Spectra of the accelerometer readings indicating the 2.52 and 2.60 Hz values as the first natural frequencies of the mast in each of the two principle directions. This analysis showed that the cause of the mast vibration was excitation by the propeller shaft rotation. The trials conducted by DREP to determine the magnitude of the excitation spectrum were not conclusive. The amplitude of excitation did not show a marked increase at any particular speed value, although there was some evidence of increased excitation in the 152 to 156 rpm range. This result could indicate a hull mode being present. In addition to the recorded data, the DREP trials officer reported experiencing significant vibration in the ship in this speed range.

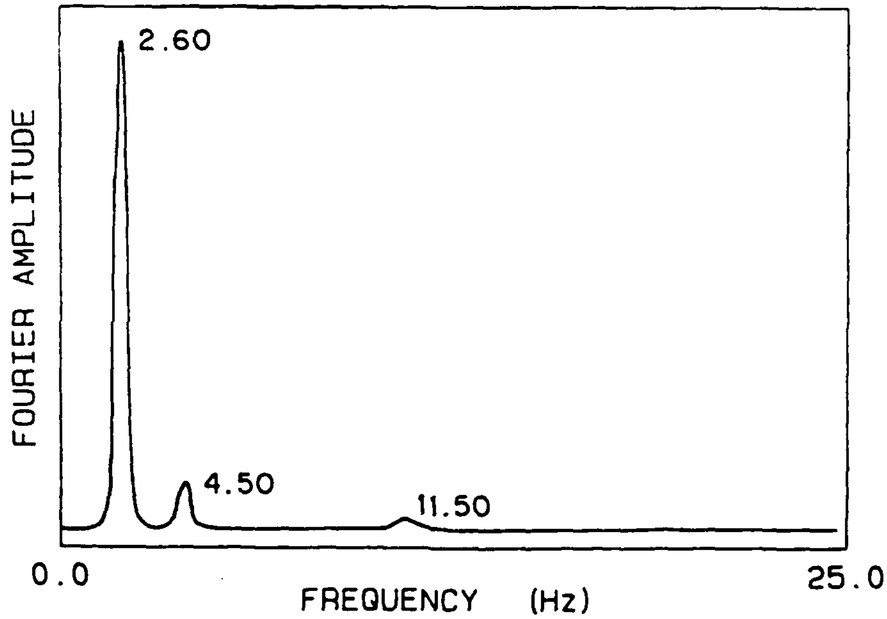

FIGURE 9: FREQUENCY SPECTRUM FROM ACCELERATION RECORD OF TRIAL 2; MAST MANUALLY SHAKEN IN ATHWARTSHIP DIRECTION, ACCELEROMETER AT STATION 1

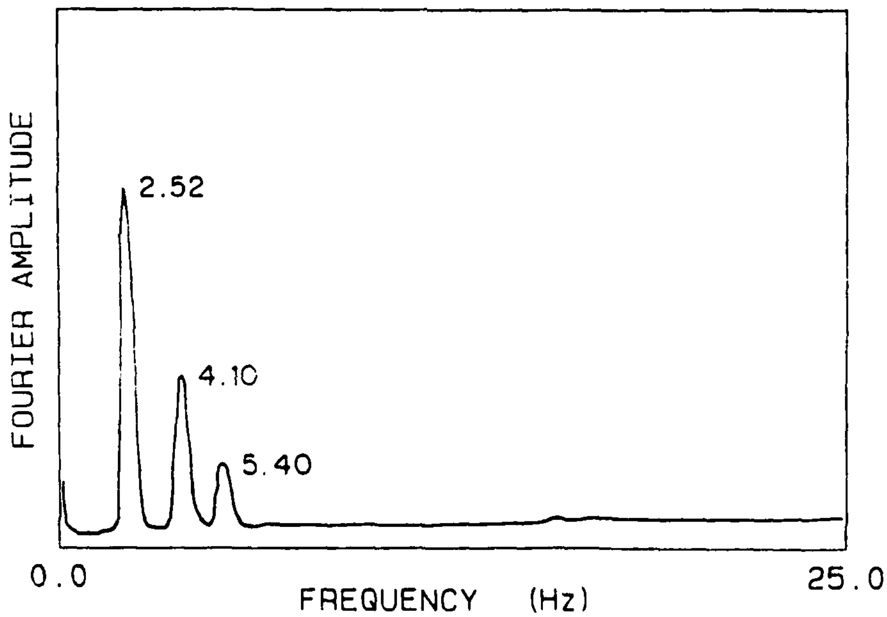

FIGURE 10: FREQUENCY SPECTRUM FROM ACCELERATION RECORD OF TRIAL 1; MAST MANUALLY SHAKEN IN FORE-AFT DIRECTION, ACCELEROMETER AT STATION 3

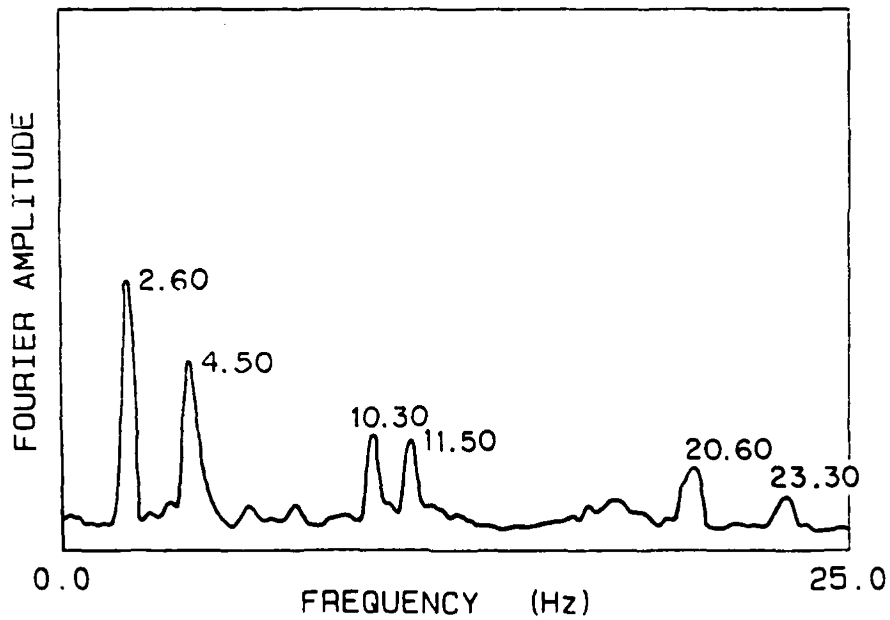

FIGURE 11: FREQUENCY SPECTRUM FROM ACCELERATION RECORD OF TRIAL 4; MAST STRUCK WITH HAMMER IN ATHWARTSHIP DIRECTION, ACCELEROMETER AT STATION 1

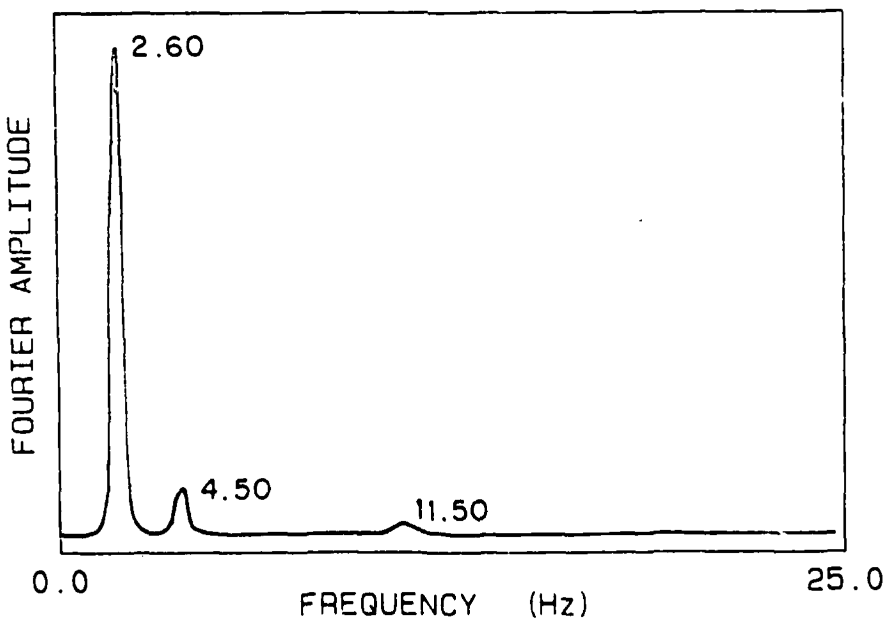

FIGURE 12: FREQUENCY SPECTRUM FROM ACCELERATION RECORD OF TRIAL 3; MAST STRUCK WITH HAMMER IN FORE-AFT DIRECTION, ACCELEROMETER AT STATION 1

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The finite element program VAST, and experimental vibration data are used to analyze a serious vibration problem with the new mast on the Mackenzie class ship, DDE 261, HMCS QU'APPELLE. It has been determined that the mast's natural frequency of vibration falls into the range of excitation frequencies produced by the propeller shaft rotation. It is postulated that the propeller excitation is driving both resonant hull and mast modes together.

After analysis of some possible solutions using VAST, it is determined that there is no simple solution that would shift the mast's natural frequency completely out of the range of excitation resulting from the propeller shaft rotation. However, some trial solutions are suggested that may provide sufficient attenuation of the mast vibration for safe and normal operation of the ship. In addition, an analysis approach is suggested which would avoid this type of problem in future designs.

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